MODELING THE FUEL BURNING RATE IN DIESEL ENGINES

William E. Remley

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by

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ABSTRACT

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Submitted to the Department of Ocean Engineering on 12 May, 1972, in partial fulfillment of the requirements for the degrees of Master of Science in Naval Architecture and Marine Engineering and Master of Science in Mechanical Engineering.

A four inch bore, open chamber, semi-quiescent diesel, in the Sloan Test Laboratory was used to study the fuel burning rate of a diesel engine. Operating conditions were adjusted to achieve cases of predominately second stage and third stage combustion. A simple mathematical model of the combustion process was constructed, assuming a perfect gas, and computerized to predict the amount of fuel burned as a function of crank angle. The measured pressure time data from the test engine was used in the computer model. It was necessary to include heat transfer effects to get acceptable agreement with the measured fuel flow rate.

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W.E. Remley

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CHAPTER 1

INTRODUCTION

The recent public demands for clean air have resulted in a need for increasingly better control of air pollution from all sources. In the area of transportation most of the attention has been given to spark-ignition engines. As the development of control methods for emissions from spark-ignition engines progresses, the contribution of diesel-powered vehicles to the air pollution problem will become more significant unless diesel emissions are reduced. With respect to future clean air requirements, many diesel engines are either low or moderate with regards to carbon monoxide (CO) and hydrocarbon (HC), but high with respect to the oxides of nitrogen (NO_x). Hence, the need for control of diesel emissions applies primarily to controlling the NO_x without increasing the other emissions. (CO, HC, smoke, or odor)

Thus this study was undertaken as part of a more extensive program to examine and model the formation of NO_x in diesel engines. Combustion in diesel engines is usually thought of as taking place in three stages. The first of these is called the delay period or ignition lag when there is a mixing of evaporated fuel and fuel droplets with the air. The second stage occurs with the rapid burning of this

premixed fuel and air. The third stage is a result of the fuel that has not burned, together with fuel subsequently injected, burning at a controlled rate governed by the availability of oxygen necessary for combustion. Due to the "premixed" and "mixing controlled" burning, products of different fuel to air ratios result which affect the nitrogen oxide formation process. The objective of this study, then, was to simulate as closely as possible a case of predominately second stage combustion and a case of predominately third stage combustion with a test diesel engine. Having satisfactorily achieved the desired combustion cases the amount of nitrogen oxides formed during each case was to be measured and the model correlated with the experimental data.

In addition to simulating certain combustion states a simple combustion model was formulated. The computerized version of this model calculates the mass fraction of fuel burnt as a function of the crank angle, as well as a burnt gas temperature for a fully "mixed case" and individual element temperatures for an "unmixed case". The "mixed case" assumes infinite heat transfer and uniform mixing of the burnt gases and computes a mean burnt gas temperature. The "unmixed case" assumes no heat transfer and no mixing and computes the temperature of an element assuming it is isentropically compressed after it is burned.

Using well known theory on the behavior of diesel

combustion in relation to inlet conditions, injection timing, and fuel characteristics, the cases of predominately third and second stage combustion were readily achieved. That is, a high inlet temperature and pressure, with late injection and a low injection rate shortens the delay for predominately third stage combustion and early injection with a high injection rate causes a longer delay for predominately second stage combustion.

The computer model was developed using the most representative cases and their associated pressure time data as input.

CHAPTER II

APPARATUS AND PROCEDURE

A. THE ENGINE

The engine used in this study was a 4 inch bore, 2½ inch stroke, open chamber, semi-quiescient diesel in the Sloan Automotive Laboratory. A hemispherical piston used in earlier experiments by HAMILTON (2) and DUNCAN and LOGTERMAN (1) was left in the engine and used during this study. A list of engine specifications appears in Table II-1.

B. FUEL INJECTION SYSTEM

The fuel system consisted of a Bosch, single plunger, fuel pump and a Roosa Master fuel nozzle. Injection timing was set by means of an adjustable coupling between the engine cam shaft and the pump drive shaft. The engine drive utilized was the same one driving the valve cams so that there was a 2:1 speed reduction.

Actual injection advance was determined by observing the beginning of injection from a nozzle with a strobelight. The strobelight was triggered by a signal from automotive breaker points fitted on the engine crankshaft drive. The injection was observed in a glass jar and the angle read on the engine's flywheel.

TABLE II-1

ENGINE SPECIFICATIONS

No.	Item	
1.	bore	4.0 in
2.	stroke	2.5 in
3.	cylinder displacement	31.41 in^3
4.	conn rod length	6.25 in
5.	compression ratio	14.3:1
6.	number of compression and oil rings	2,1
7.	number of inlet and exhaust valves	2,2
8.	valve diameter	1.286 in
9.	valve lift	0.280 in
10.	inlet valve timing open/closes	15°BTC/50°ABC
11.	exhaust valve timing open/closes	50°BBC/15°ATC
12.	diameter of intake manifold pipe	2.00 in
13.	diameter of exhaust manifold pipe	1.60 in
14.	type of engine	4 stroke

TABLE II-2

FUEL INJECTION SPECIFICATIONS

No.	Item	
1.	number of nozzle sprays	8
2.	diameter of nozzle holes	0.008 in
3.	length of nozzle holes	0.020 in
4.	angle of spray from horizontal	7°
5.	length of nozzle, inlet stud to tip	3.65 in
6.	diameter of nozzle body	0.375 in
7.	I.D. of inlet stud	0.100 in
8.	nozzle cracking pressure	2500 psi
9.	length of injection line	24 in.
10.	I.D. and O.D. of injection line	0.062, 0.250 in
11.	fuel pump model identification	APE/1B 80P300/3-5671 Serial #269347
12.	injection pump plunger diameter	5 mm.
13.	max. plunger lift	0.3860 in

The Roosa nozzles are very compact and lend themselves to being installed in the four valve engine head. The nozzle opening pressure was set at 2500 psi. Table II - 2 lists the fuel injection specifications.

C. COMPRESSED AIR SYSTEM

The air compressor in the pump room of the Sloan Laboratory was used to supply supercharging air. The compressor discharge goes to a storage tank with a pressure regulating valve which regulates the lab mains to within 2 psi at a nominal 100 psi. Further reduction occurred in the test cell so that the final supercharging air fluctuated only about 0.8 inches Hg at the engine.

D. INSTRUMENTATION

(1) Fuel Flow

Fuel consumption was measured gravimetrically using the system shown in Figure II - 1. The system's only outlet is the fuel injected into the engine; all other lines return to the glass beaker. Additional fuel is added to the beaker between fuel measuring cycles by drawing from the fuel tank. The time required to use a specified weight of fuel is measured by an electrically tripped timer. The booster pump was used to supply a positive suction pressure to the Bosch Fuel Pump.

(2) Air Flow

The engine's air consumption was measured by an

INSTRUMENTATION OF FUEL SYSTEM

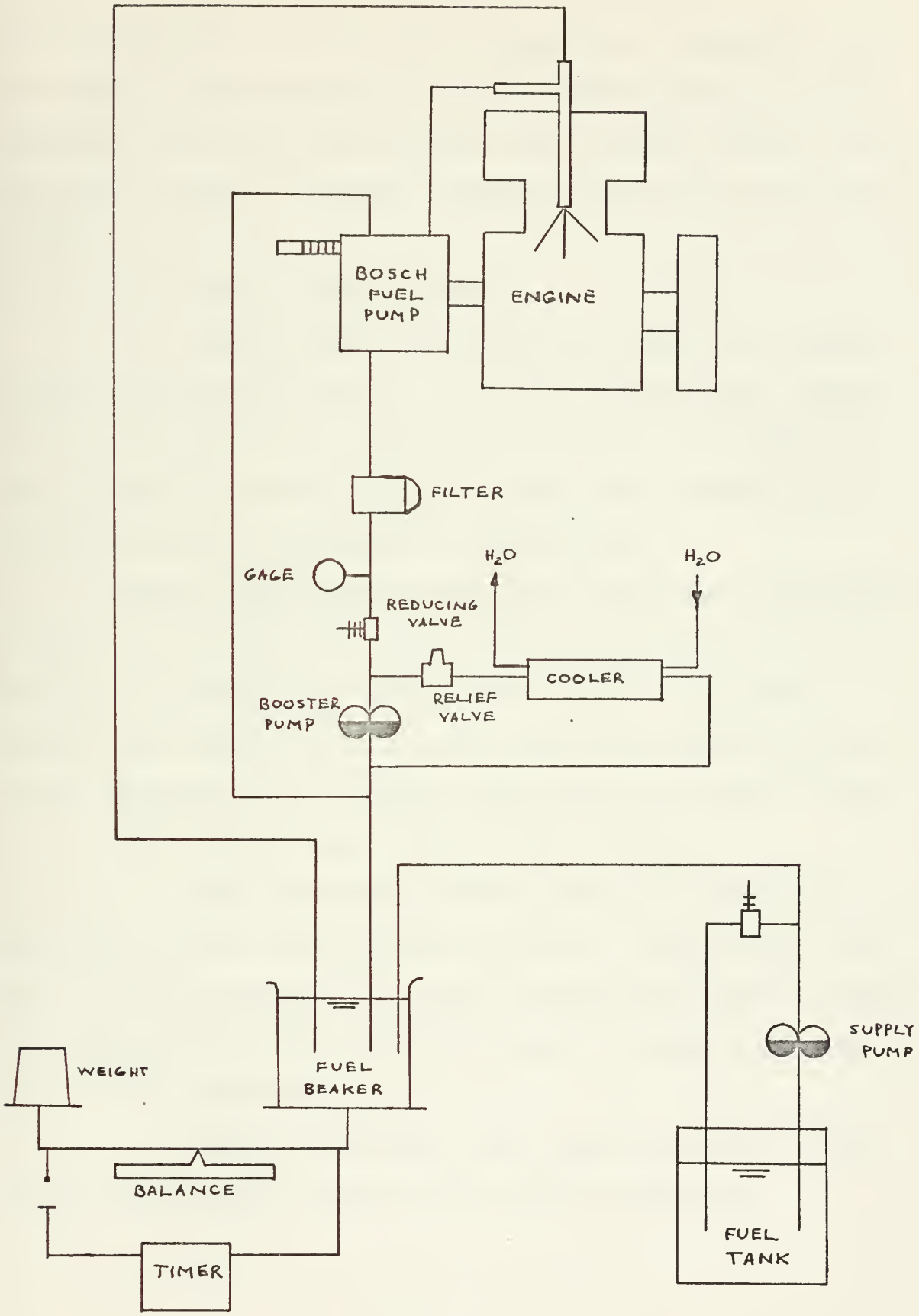


FIGURE II - 1

ASME square-edged orifice with flange taps. Manometers were connected to these taps to indicate static pressure and the pressure drop across the orifice. The orifice diameter used was 0.614 inches. Appendix A gives the air flow calibration curves.

(3) Engine Power and Speed

Engine power was measured by means of a direct current dynamometer driven at one half engine speed. Brake load is transmitted to a hydraulic cylinder which reads out as a column of mercury in a manometer. The constant 3.15 converts inches of mercury into BMEP in psi.

Engine speed was measured by a mechanical tachometer driven off the fuel pump shaft. This necessitated that all readings be doubled for actual engine speed. To insure steady speed during a run, an on line stroboscope is aimed at the flywheel where markings come into view every 25 RPM.

(4) Temperatures and Water Flow Rates

Iron-constantan thermocouples are installed for measuring oil and water, inlet and outlet temperatures, and the air inlet temperature. These thermocouples send a signal to a dial potentiometer where they can be selectively read.

(5) Pressures

Mercury manometers were used to measure inlet, exhaust, and static upstream air orifice pressures.

(6) Engine Combustion

An M.I.T. balanced-pressure type indicator was used to obtain a trace of the pressure time history in the cylinder. Nitrogen was used as a pressure source to balance the pressure in the cylinder. A pressure sensing device located in the head of the engine and opening into the combustion chamber sends a signal to the trip circuit of the indicator. A diaphragm in the pressure sensing device makes or breaks an electrical circuit which causes a spark to be generated by the trip circuit of the indicator. This spark makes a mark on a drum rotating at engine speed. The spark is positioned by the nitrogen pressure source which acts on a plunger constrained by a spring. The marks on the rotating drum correspond to the pressure present in the combustion chamber at that particular crank angle. The indicator has additional circuits for marking the top dead center position and a reference pressure on the drum.

E. TEST PROCEDURE

(1) Data Sought and Recorded

The objective of the tests was to obtain a case of predominately third stage combustion and a case of predominately second stage combustion. With each run to achieve these results a full set of engine data was taken. This consisted of fuel consumption, air consumption, air inlet temperature, oil inlet temperature, head and barrel

inlet temperatures, exhaust temperature, air inlet and exhaust pressures, engine BMEP and FMEP, and an indicator card.

For the two cases the conditions that were to be duplicated as closely as possible were a lean F/A ratio, approximately one-half stoichimetric, and equal power output.

The following approach was adopted before the actual testing was attempted: For predominately third stage combustion a small plunger was to be used in the fuel pump to obtain a long injection angle. The engine was to be run at low RPM for low injection rate, and the injection angle, or duration of injection from beginning to end, was to be such that it was almost split by the top center position of the piston, the main objective being to reduce the delay time sufficiently so that the slow controlled burning characteristic of third stage combustion could be achieved.

For predominately second stage combustion a larger diameter plunger was to be used in the fuel pump and the engine run at a higher RPM to achieve a high injection rate. The injection duration was to be short and occur before the piston reached top dead center position. The objective being to increase the delay time so that the fuel and air were well mixed when combustion occurs.

(2) Procedure

Testing started on the case of predominately

third stage combustion. Injection timing was set as specified and the smallest commercially available plunger was installed in the Bosch Pump. The immediate results were indicator cards showing a rapid pressure rise and long delay time, more indicative of second stage than third stage burning. In order to further reduce delay time the inlet air temperature was increased and supercharging was added to raise the inlet pressure. These measures were very effective in reducing the delay time. The indicator cards of runs made under these conditions of high inlet pressure and temperature show burning soon after injection and a gradual uniform pressure rise.

Numerous tests were run at the same conditions to insure equilibrium of all parameters. Normally each test run lasted for approximately fifteen minutes.

Friction readings were taken at the end of the firing tests while the engine temperatures were nearly the same as during running.

Several factors contributed to not attempting a special series of tests for predominately second stage combustion. Firstly, since the plunger used for the third stage combustion required a small rack setting (short effective plunger stroke) it was felt that a larger diameter plunger might result in not being able to obtain the proper fuel flow rate, or with the existing fuel supply system a very small rack setting would result in poor injection performance.

Thus the data available representing cases of predominately third and second stage combustion were used to develop the computer model.

CHAPTER III

COMBUSTION MODEL

To analyze the data taken from the test engine a mathematical model was developed, using the approach taken by LAVOIE, et. al. (4) in their studies on nitrogen oxide formation in spark ignition engines. At the temperatures and densities in the compression ignition engine, it is reasonable to assume the volume of the reaction zone as negligible and that the gas within the cylinder consists of a burned fraction at thermodynamic equilibrium plus an unburned fraction composed of fuel vapors and air. In addition if we assume that the pressure is uniform throughout the cylinder and that the fuel vapor, air, and burnt gases behave as perfect gases over the range of interest their conditions are determined by the perfect gas relationships.

Thus the equations of state for the gases are:

$$pv_b = R_b T_b \tag{1}$$

$$pv_a = R_a T_a \tag{2}$$

$$pv_f = R_f T_f \tag{3}$$

where the subscripts b, a, and f are for burnt gases, air and fuel vapors respectively.

At any instant the cylinder volume, V, must be equal to the volume of its constituents or:

$$V = V_a + V_f + V_b \tag{4}$$

Since energy is to be conserved

$$E_o - Q - W = E_a + E_f + E_b \tag{5}$$

Writing equations (4) and (5) in terms of the respective masses of the cylinder gases at any instant

$$V = v_a m_a + v_f m_f + \int_o^{m_b} v_b dm_b \tag{6}$$

and

$$E_o - Q - W = e_a m_a + e_f m_f + \int_o^{m_b} e_b dm_b \tag{7}$$

The mass relationship for the cylinder contents being:

$$m_o = m_a + m_f + m_b \tag{8}$$

The internal energy relationships are, assuming constant specific heats for all the components of the cylinder gas

$$e_f = Cv_f T_f + h^o_f \tag{9}$$

$$e_b = Cv_b T_b + h^o_b \tag{10}$$

$$e_a = Cv_a T_a + h^o_a \tag{11}$$

Substituting equations (1), (2), and (3) into (6) and (7) gives a resulting equation of state in terms of the cylinder components.

$$PV = R_a T_a m_a + R_f T_f m_f + R_b \bar{T}_b m_b \tag{12}$$

and

$$E_o - Q - W = (Cv_a T_a + h^o_a)m_a + (Cv_f T_f + h^o_f)m_f + (Cv_b T_b + h^o_b)m_b \tag{13}$$

$\bar{T}$ is the mean temperature of the burned gas if it is uniformly

mixed. However, this may not always be so. By further specifying the fuel to air ratio of the burnt gases as F_b and the mean fuel to air ratio as F_o, equations (12) and (13) can be solved to obtain:

$$ym_{fo}/F_b = \Big\{\big[PV - P_oV_o + (\gamma_b - 1)(W + Q) + m_{fo}C_{vf}(\gamma_b - \gamma_f)(T_f - T_{fo})\big] + m_{ao}C_{va}(\gamma_b - \gamma_a)(T_a - T_{ao})\Big\} \Big/ \Big\{F_bC_{vf}(\gamma_b - \gamma_f)T_f + C_{va}(\gamma_b - \gamma_a)T_a + (\gamma_b - 1)\big[h_{ao} + F_bh_{fo} - (1 + F_b)h_{bo}\big]\Big\} \quad (14)$$

And

$$R_b\bar{T}_b = (R_aT_a + F_bR_fT_f) / (1 + F_b) + F_b\,pVF_o - m_{fo}(R_aT_a + F_oR_fT_f) / m_{fo}F_o(1 + F_b)y \quad (15)$$

F_b is not taken as equal to F_o since the actual burning takes place at a fuel to air ratio other than F_o. In addition F_b is assumed constant assuming a "mixing controlled flame" whose products should correspond nearly to stoichiometric conditions. y is the mass of fuel burnt and $\gamma = C_p/C_v$ and o refers to any convenient reference state such as the beginning of injection. By definition then, at the start of fuel injection:

$$P_oV_o = R_aT_{ao}m_{ao} + R_fT_{fo}m_{fo} \quad (16)$$

and

$$W = \int_{vo}^{v} pdV' \quad (17)$$

Assuming the unburnt portions of the cylinder gases to be initially uniform and subject to an isentropic compression then:

$$T_a/T_{ao} = (p/p_o)^{(\gamma_a - 1)/\gamma_a} \tag{18}$$

And

$$T_f/T_{fo} = (p/p_o)^{(\gamma_f - 1)/\gamma_f} \tag{19}$$

With equations (14), (15), (18), and (19) the quantities y and $\bar{T}_b$ can be determined from the thermodynamic properties of the burned and unburned gases and measured values of p, v, m_{fo}, m_{ao}, F_o, T_{fo}, and Q.

To proceed further some assumptions must be made about conditions in the burned gas. Having already considered the case where the temperature of the burned gas is uniform so that $T_b = \bar{T}_b$, the state of the burned gas is then given by equations (1), (14), (15), (18), and (19). This case assumes infinite heat conductivity in the burned gas. If it is also assumed that the burned gas composition is uniform, this would correspond to a "fully mixed" model and is representative of one extreme.

Having considered a case for uniform mixing of the burned gases the opposite would be a case without mixing of the burned gases. For this case individual elements of burnt gas are considered and their temperature is determined assuming they are isentropically compressed after combustion such

that:

$$T_b(y',y)/T_b(y') = \left[P(y)/P(y')\right]^{(\gamma_b - 1)/\gamma_b} \qquad (20)$$

$T_b(y',y)$ is the temperature of the element which burned at the pressure $P(y')$ when the pressure is $P(y)$. And:

$$T_b(y') = \left\{C_{pa}T_a(y') + FC_{pf}T_f(y') + h_{ao} + F_b h_{fo} - h_{bo}(1 + F_b)\right\} / C_{pb} \qquad (21)$$

is the temperature resulting from isenthalpic combustion of the gas at the pressure $p(y')$. This case assumes zero heat conductivity in the burned gas, and no mixing. This case then represents an "unmixed" model, or the opposite extreme of the "fully mixed" case. It should be noted that the "fully mixed" model includes the effect of heat transfer losses and hence $\bar{T}_b$ will be affected while the "unmixed model" does not consider these losses because it is an isentropic compression.

Appendix B gives a detailed description of the computer program.

CHAPTER IV

EXPERIMENTAL RESULTS AND DISCUSSION

This chapter presents the results obtained when the pressure time curves from the test engine were used in the computerized combustion model. Figure IV - 1 shows the pressure time diagram from run A which was taken as a case of predominately third stage combustion. Figure IV - 2 shows the pressure time diagram from run B which was taken as a case of predominately second stage combustion.

Run A met the conditions specified for third stage combustion very well. That is, the delay time was short, the duration of injection started before top dead center, and lasted beyond this point, and burning started soon after injection before extensive mixing of the air and fuel occurred. Run B, though it did occur while trying to obtain third stage combustion, met two of the prescribed parameters for second stage combustion. Namely, the delay time was long, giving the fuel and air time to mix thoroughly before combustion occurred, and a shorter injection period was used. Tables IV - 1 and IV - 2 list the conditions of operation for runs A and B.

The first set of computer calculations did not include heat transfer effects. The output of this program in terms of mass fraction of fuel burnt (Y) versus crank angle is shown

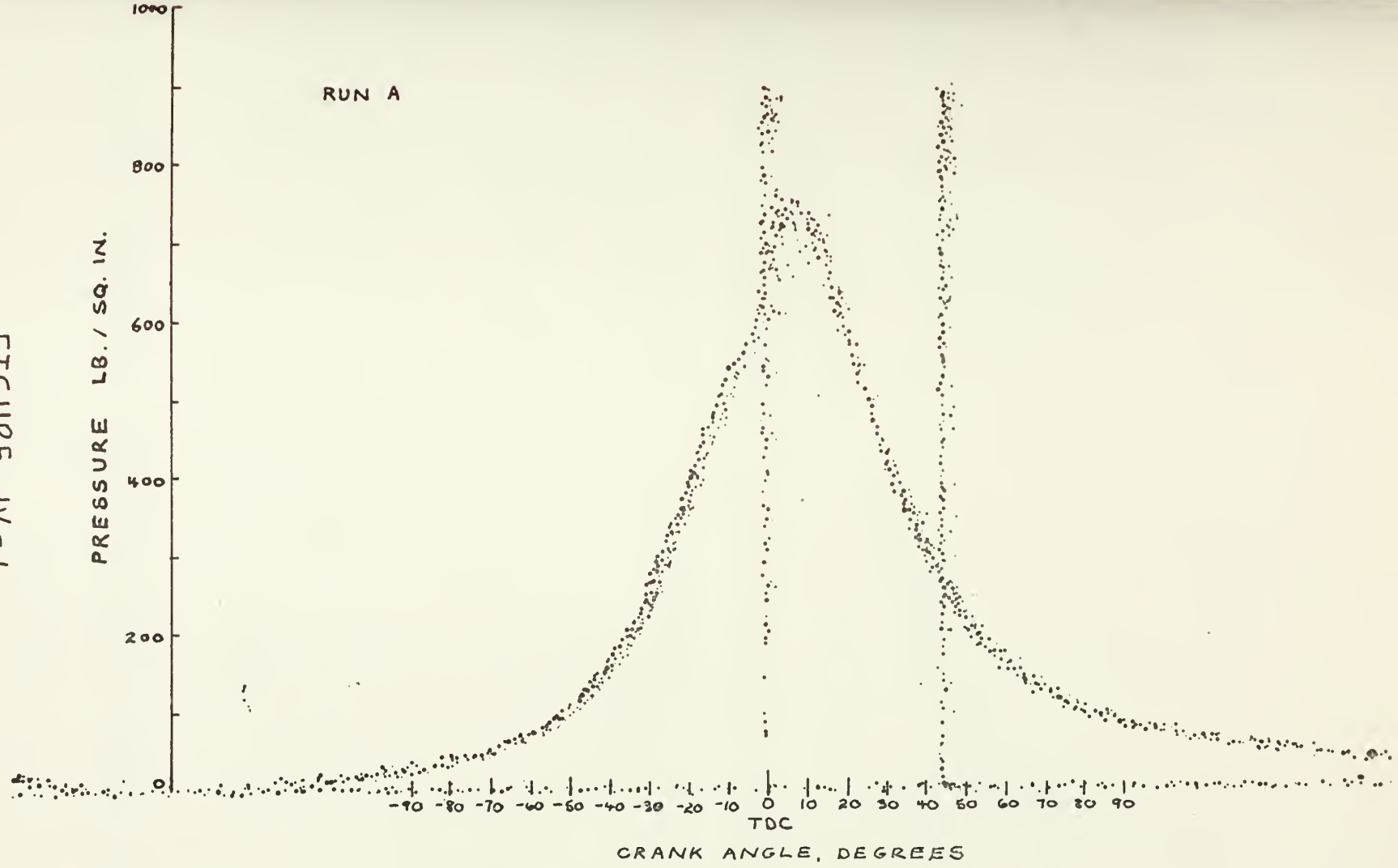


FIGURE IV-1

PRESSURE-CRANK-ANGLE DIAGRAM OF RUN A
PREDOMINANTLY THIRD STAGE COMBUSTION

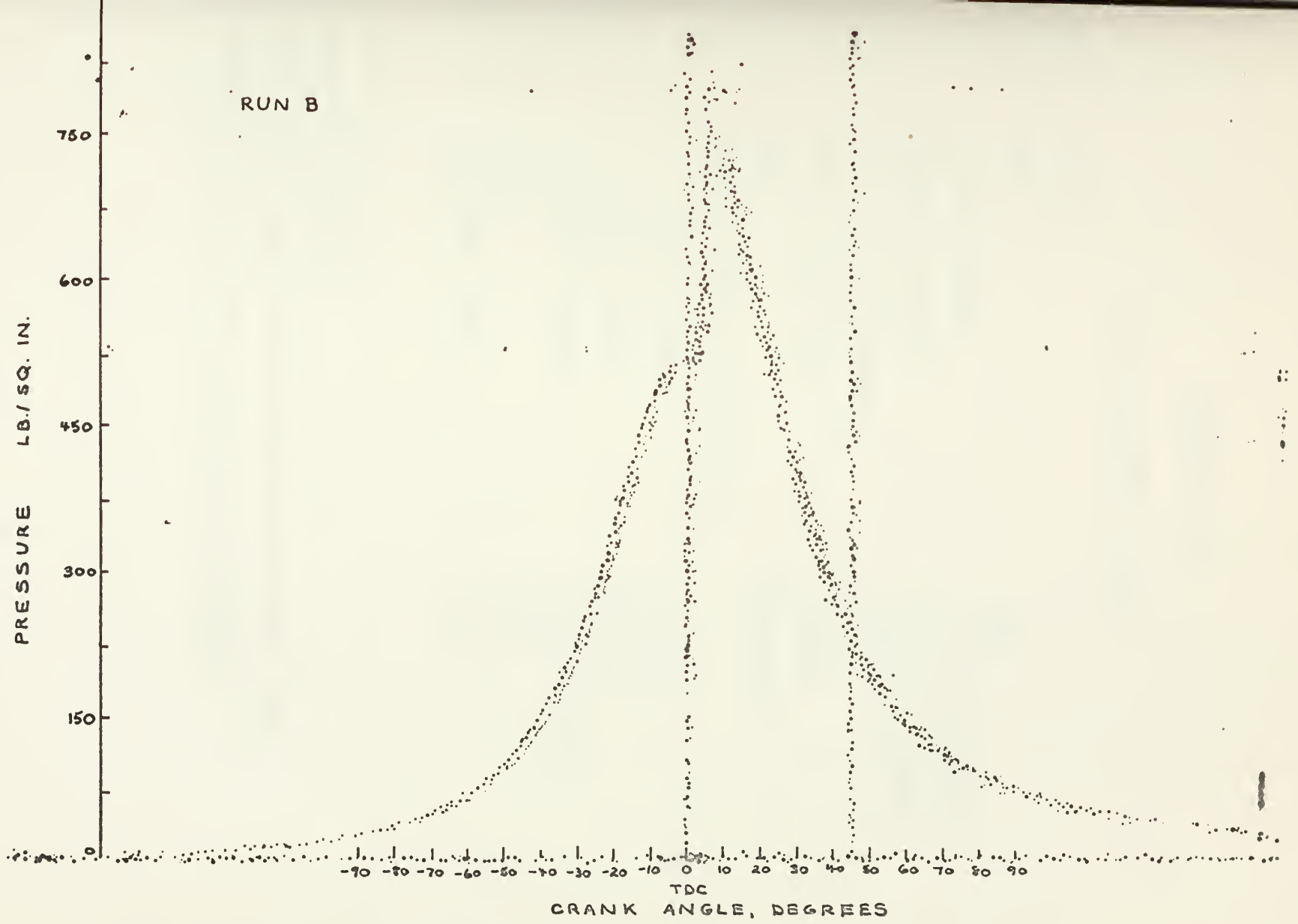


FIGURE IV-2

PRESSURE-CRANK-ANGLE DIAGRAM OF RUN B
PREDOMINANTLY SECOND STAGE COMBUSTION

TABLE IV - 1

DATA COLLECTED - RUN A

No.	Item	
1.	RPM	1450
2.	Inlet pressure	13.5 in. Hg gage
3.	Exhaust pressure	13.4 in. Hg gage
4.	Inlet air temperature	186°F
5.	Fuel to air ratio	0.0394
6.	IMEP	88.1 psi
7.	Cooling water temperatures	
	a. Barrel (inlet)	190.5°F
	b. Head (inlet)	204.0°F
8.*	Injection advance	12.5° B.T.D.C.
9.**	Delay	5.5°
10.***	Injection angle	17.5°

*Start of injection.

**Combustion begins as indicated on indicator card.

***Duration of injection from start to finish.

TABLE IV - 2

DATA COLLECTED - RUN B

No.	Item	
1.	RPM	1290
2.	Inlet pressure	10.6 in. Hg gage
3.	Exhaust pressure	Atmospheric
4.	Inlet air temperature	187°F
5.	Fuel to air ratio	0.0319
6.	IMEP	75.5 psi
7.	Cooling water temperatures	
	a. Barrel (inlet)	187.0°F
	b. Head (inlet)	200.5°F
8.*	Injection advance	14.0° B.T.D.C.
9.**	Delay	17°
10.***	Injection angle	15°

*Start of injection.

**Combustion begins as indicated on indicator card.

***Duration of injection from start to finish.

in Figure IV - 3. Since both run A and B were made with lean fuel to air ratios the results are in error, as Y should equal one when the fuel is fully burned. However, qualitatively the appearance of the curves (Figure IV - 3) is practically what you would expect. The curve for fuel burnt versus crank angle of run A is a smooth curve with a steep slope at first then tapering off, indicating the presence of a predominately controlled burning rate. On the other hand the Y versus crank angle curve for run B is not so smooth. Its initial slope is very steep, almost vertical, then changes to a more gradual slope, indicating a predominately rapid burning rate turning into a controlled rate of burning after the premixed fuel and air was consumed.

Research on the subject of heat transfer in diesel engines led to the belief that heat transfer effects should be considered. (5,6,7) Several methods for calculating the heat transfer between the cylinder gases and the wall of the combustion chamber were examined. But all the proposed methods examined for calculating heat transfer in diesel engines relied on experimental data such as the instantaneous wall temperature or the instantaneous heat flux at a point in the cylinder. Such data was not available on the test diesel used in this study. To observe the effects of heat transfer on the magnitude of Y a plot of total heat flux as a function of crank angle was taken from research by

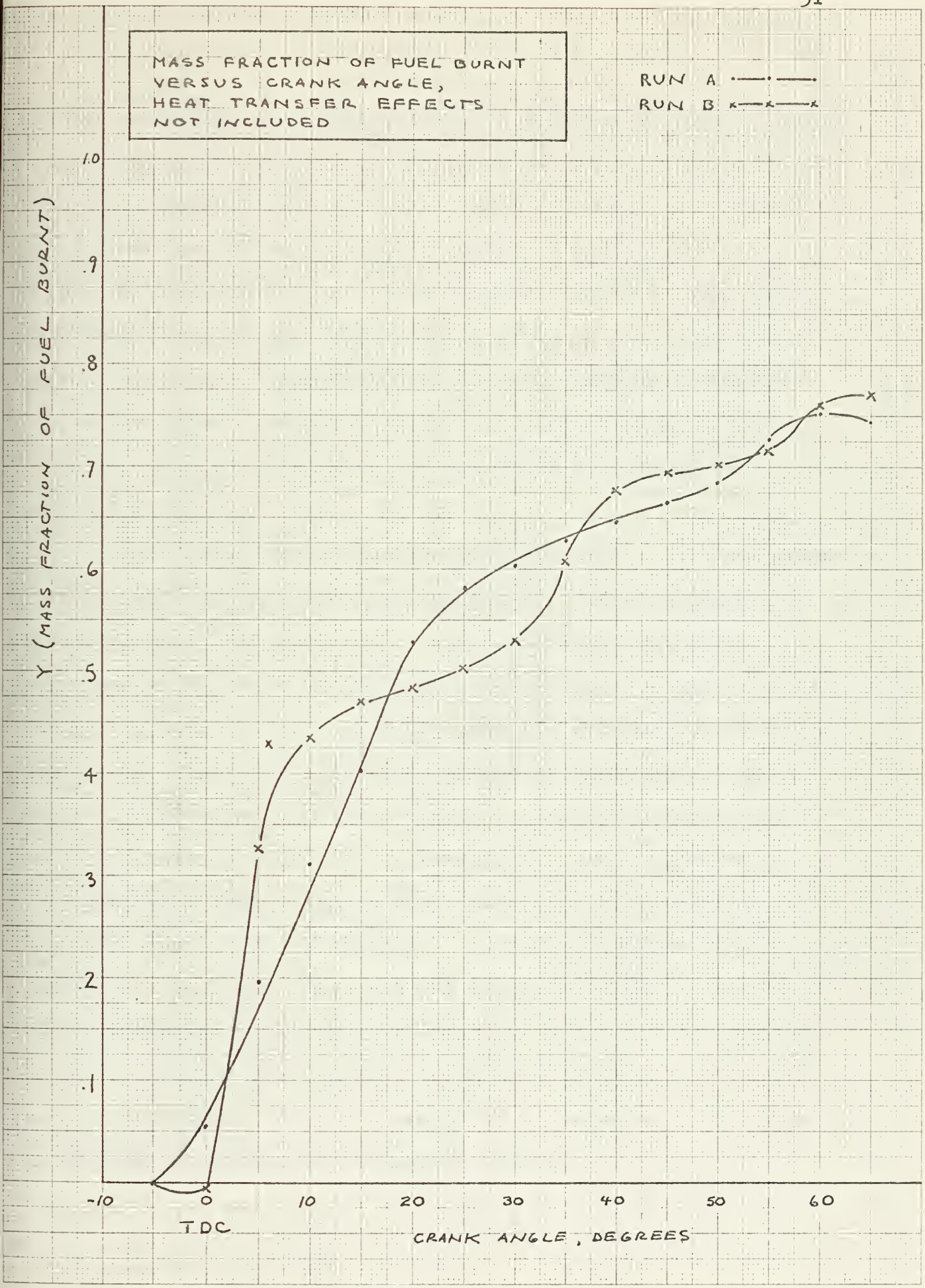

MASS FRACTION OF FUEL BURNT VERSUS CRANK ANGLE, HEAT TRANSFER EFFECTS NOT INCLUDED
RUN A
RUN B
Y (MASS FRACTION OF FUEL BURNT)
1.0
.9
.8
.7
.6
.5
.4
.3
.2
.1
-10
0
10
20
30
40
50
60
TDC
CRANK ANGLE, DEGREES

FIGURE IV - 3

OGURI and INABA. (7) This particular diagram was from a diesel operating at 900 RPM and producing 91 psi IMEP. Figure IV - 4 shows the heat flux versus crank angle for these conditions. This heat flux curve was scaled so that Y was forced to go to 1.0. For run A the flux values were multiplied by the factor 2.9 and for run B they were multiplied by 2.3 to force Y to 1.0. These values were then varied by $\pm$ 25% and related values of Y were computed. Curves showing Y equal to 1.0 and with heat transfer values varied by $\pm$ 25% are shown in Figures IV - 5 through IV - 7.

To justify this approach of forcing Y to 1.0 the entire scaled heat flux versus crank angle curve was integrated to give the heat lost per cycle. This integrated value was compared with a best estimate of the heat lost to the engines cooling system. Since heat loss data had not been recorded during the test runs a previous thesis by HAMILTON (2) was used to extrapolate a best estimate of the heat lost to the cooling system. Table IV - 3 summarizes this comparison. The best correlation occurs when the scaled heat flux for Y equal to one is reduced by 25%. For this situation the maximum value of Y is 0.945 for run A and 0.964 for run B. This is a considerable increase above the maximum values of Y which were computed without heat transfer effects. Namely, 0.754 for run A and 0.770 for run B. Hence, it is necessary to include heat transfer effects for accurate results.

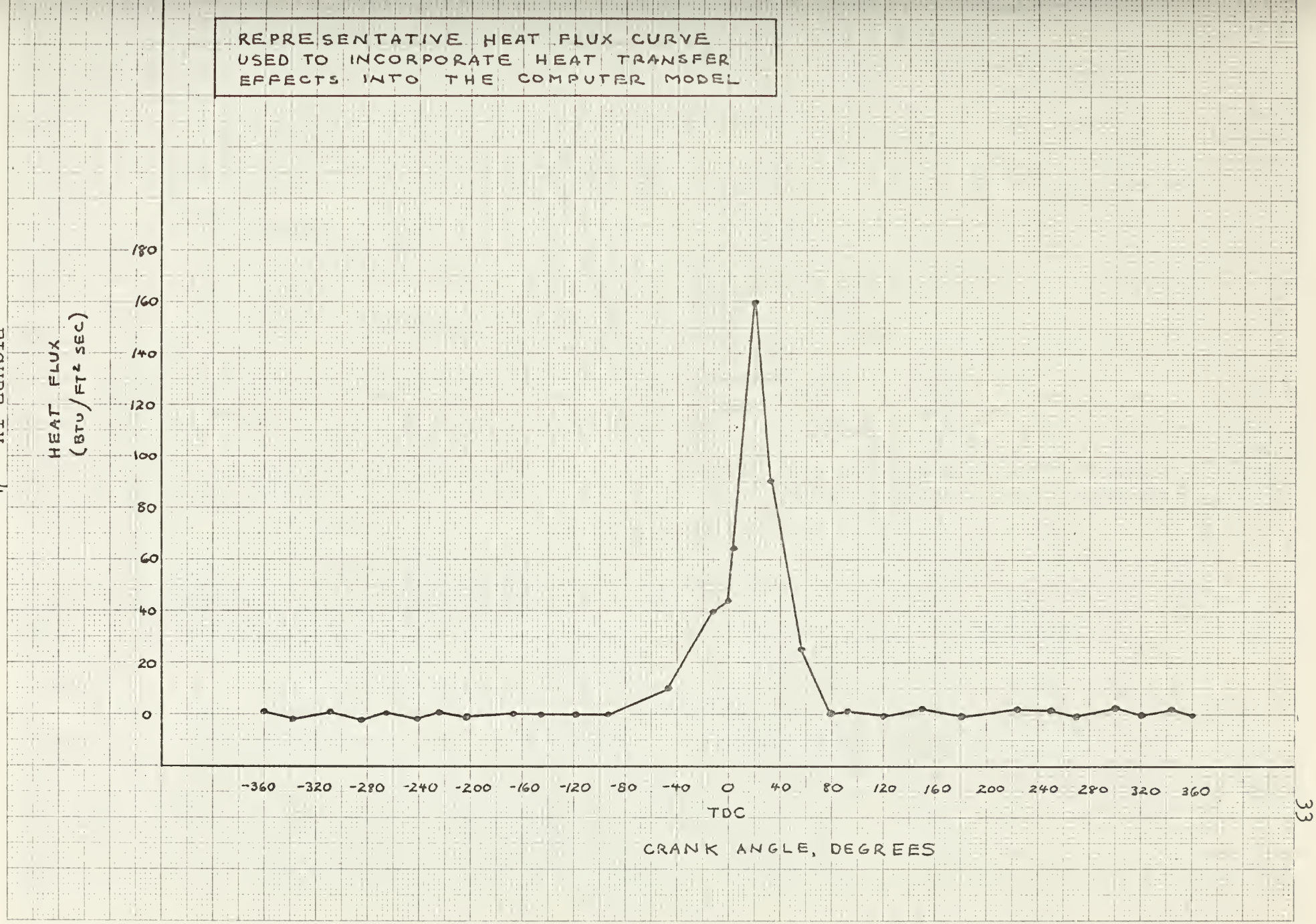

REPRESENTATIVE HEAT FLUX CURVE
USED TO INCORPORATE HEAT TRANSFER
EFFECTS INTO THE COMPUTER MODEL
HEAT FLUX
(BTU/FT² SEC)
180
160
140
120
100
80
60
40
20
0
-360 -320 -280 -240 -200 -160 -120 -80 -40 0 40 80 120 160 200 240 280 320 360
TDC
CRANK ANGLE, DEGREES

FIGURE IV - 4

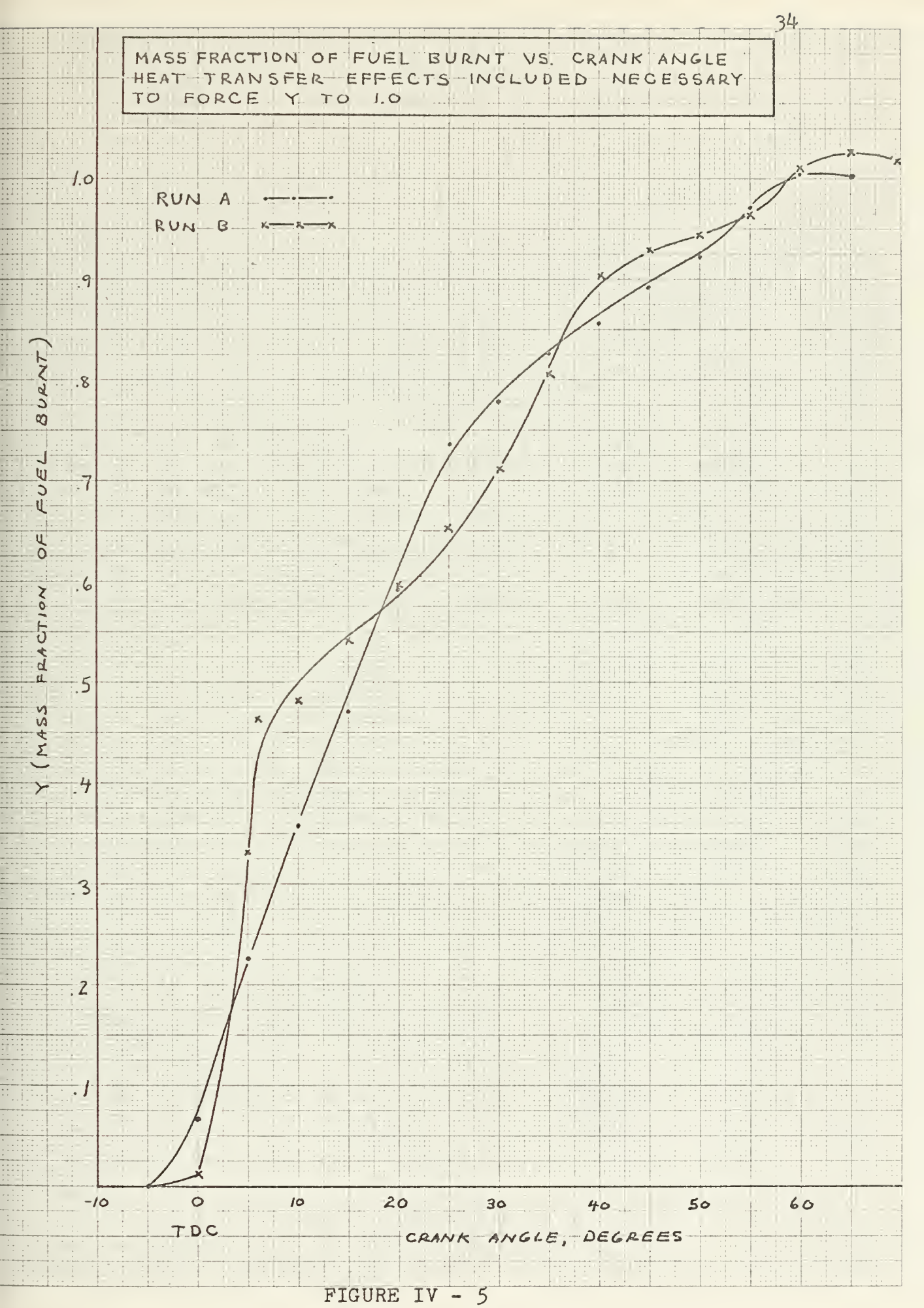

MASS FRACTION OF FUEL BURNT VS. CRANK ANGLE
HEAT TRANSFER EFFECTS INCLUDED NECESSARY
TO FORCE Y TO 1.0
RUN A
RUN B
Y (MASS FRACTION OF FUEL BURNT)
1.0
.9
.8
.7
.6
.5
.4
.3
.2
.1
-10
0
10
20
30
40
50
60
TDC
CRANK ANGLE, DEGREES

FIGURE IV - 5

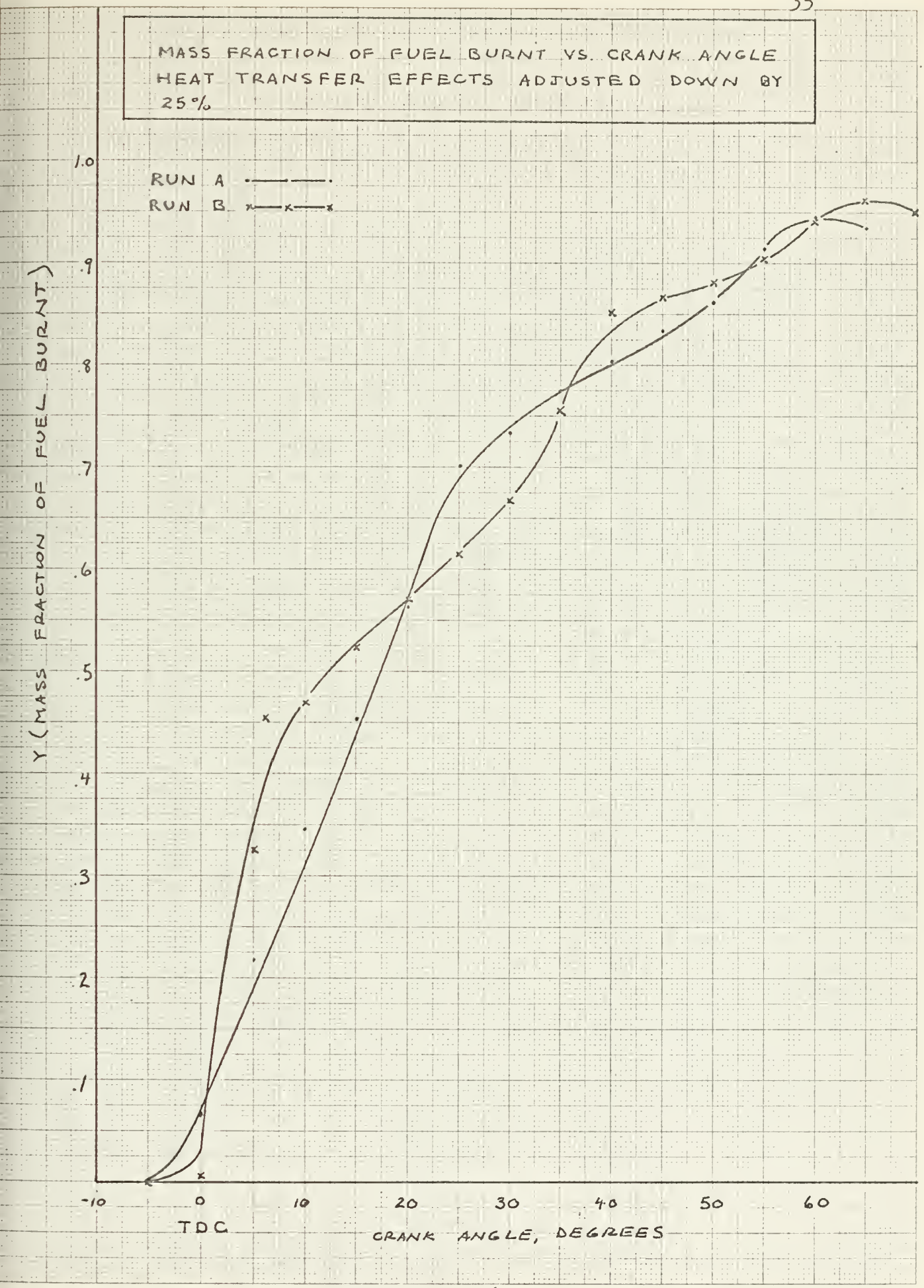

MASS FRACTION OF FUEL BURNT VS. CRANK ANGLE
HEAT TRANSFER EFFECTS ADJUSTED DOWN BY 25%
RUN A
RUN B
Y (MASS FRACTION OF FUEL BURNT)
1.0
.9
.8
.7
.6
.5
.4
.3
.2
.1
-10
0
TDC
10
20
30
40
50
60
CRANK ANGLE, DEGREES

FIGURE IV - 6

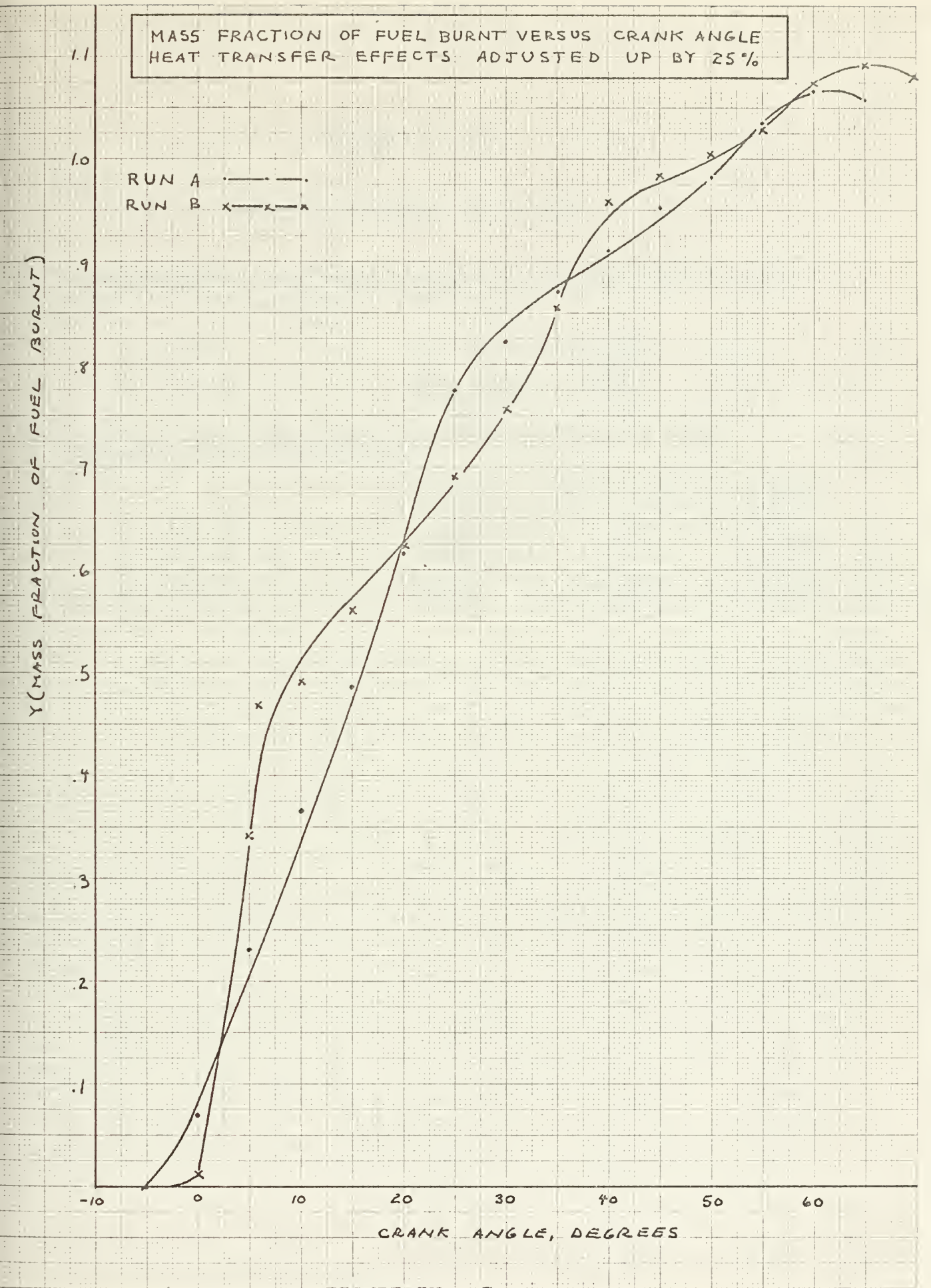

MASS FRACTION OF FUEL BURNT VERSUS CRANK ANGLE
HEAT TRANSFER EFFECTS ADJUSTED UP BY 25%
RUN A
RUN B
Y (MASS FRACTION OF FUEL BURNT)
1.1
1.0
.9
.8
.7
.6
.5
.4
.3
.2
.1
-10
0
10
20
30
40
50
60
CRANK ANGLE, DEGREES

FIGURE IV - 7

TABLE IV - 3

HEAT TRANSFERRED PER CYCLE

1. Best estimate from pervious data of the heat lost to the engine's cooling system. (BTU/cycle)

   a. Run A — heat loss = 0.309

   b. Run B — heat loss = 0.279

2. Computed heat loss from integration of scaled heat transfer curve. (BTU/cycle)

	$\dot{Q}$ -25%	$\dot{Q}$ for Y=1.0	$\dot{Q}$ +25%
a. Run A	0.318	0.425	0.532
b. Run B	0.273	0.364	0.457

The heat transfer model was incorporated into the computer program by means of a DO loop that integrates the curve of heat flux versus crank angle and adds these effects to the respective angles so they are reflected in the Y value.

Table IV - 4 shows the results of a series of computer runs made to determine the effect on Y when Δ H and the specific heat ratio of the burned gases (γ_b) were varied consistently. It is quite apparent from observing the table that there is little change in the value of Y when ΔH and γ_b are varied consistently. Section C of Appendix B explains how ΔH and γ_b are obtained.

Figure IV - 8 shows a plot of the temperatures computed for the "fully mixed" and "unmixed" cases. These values were computed in conjunction with the heat transfer flux used to bring Y to 1.0. $\bar{T}_b$ is the mean temperature of the burned gas from the "mixed model". T_b (0,Y) and T_b (Y) are temperatures from the "unmixed" model. T_b (0,Y) is the temperature of the first element to burn and T_b (Y) is the temperature of the element burned at Y. In the model all of the heat transfer comes out of the uniformly mixed burned gas. This is why $\bar{T}_b$ is low. T_b (0,Y) and T_b (Y) do not reflect the heat transfer effects since the "unmixed" model computes them based on an isentropic compression of the burned gas and air. The

TABLE IV - 4

VARIATION OF ΔH AND γ_b FOR RUN A*

	ΔH (Joules/kgs.)	γ_b	$Y_{max.}$**
1.	3.88×10^6	1.170	0.754
2.	3.0×10^6	1.230	0.744
3.	4.2×10^6	1.156	0.753
4.	4.99×10^6	1.13	0.749

*With assumed equivalence ratio of 1.0 for burned gas.

**Heat transfer effects not included.

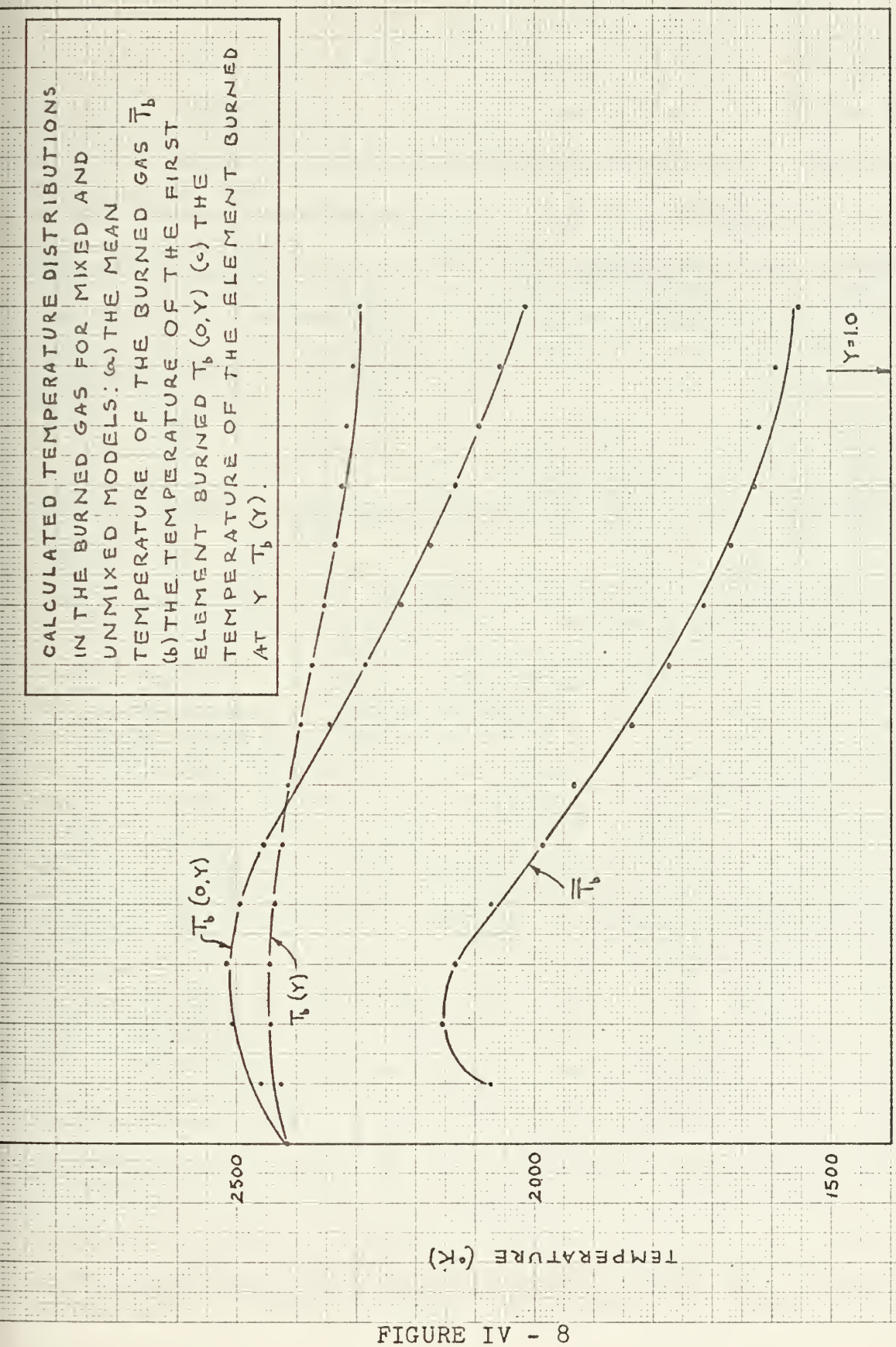

CALCULATED TEMPERATURE DISTRIBUTIONS IN THE BURNED GAS FOR MIXED AND UNMIXED MODELS: (a) THE MEAN TEMPERATURE OF THE BURNED GAS $\overline{T}_b$ (b) THE TEMPERATURE OF THE FIRST ELEMENT BURNED $T_b(0,Y)$ (c) THE TEMPERATURE OF THE ELEMENT BURNED AT Y $T_b(Y)$.
$T_b(0,Y)$
$T_b(Y)$
$\overline{T}_b$
Y=1.0
TEMPERATURE (°K)
2500
2000
1500
-5 0 5 10 15 20 25 30 35 40 45 50 55 60 65 70 75
CRANK ANGLE, DEGREES

FIGURE IV - 8

real case lies somewhere in between these two cases since there is some, but not complete, mixing of the gases involved, and not all of the burned gases are cooled uniformly. Radiation which is approximately 18% of the total heat flux cools the bulk of the burned gas and convection cools part, but not all of the burned gas substantially. (7) The cooling effect will have significant effect on the NO_x production since cooling 10% of the gas substantially will give different NO_x production than cooling all of the gas 10%.

CHAPTER V

CONCLUSIONS AND RECOMMENDATIONS

The following is a list of conclusions and recommendations which appear in the text of this report.

A. CONCLUSIONS

1. It is possible to create combustion conditions within a diesel so that two predominately different kinds of burning occur.

2. The simple mathematical model proposed gives accurate fuel burning rate predictions, when heat transfer effects are included.

3. When ΔH and gamma burnt are varied consistently, y will have the same approximate value as long as the other input parameters are held constant.

B. RECOMMENDATIONS

1. Further testing should be continued on the test diesel to achieve a case of predominately second stage combustion using a larger diameter plunger in the Bosch fuel pump and injection timing advanced beyond that of the predominately third stage case.

2. The computer model should be modified so that actual data from the test engine could be incorporated with the input for the prediction of heat transfer effects on the

fuel burning rate.

3. The test engine should be instrumented so that an accurate representation of the heat transfer effects during combustion could be constructed.

APPENDIX A

AIR FLOW EQUATIONS AND CHARTS

The air flow metering orifice equation was obtained from a report by LEARY and TSAI. (6) In its most basic form it is:

$$\dot{m} = 0.1145 D_2^2 K Y \sqrt{\frac{P_o}{T_o} G y (\Delta P)}$$

Where:

$\dot{m}$ = flow rate, lb/sec

D_2 = orifice diameter, in

K = flow coefficient, dimensionless

Y = expansion factor, dimensionless

P_o = static pressure upstream of orifice, in. Hg ABS

T_o = temperature at orifice, ^oR

G = specific gravity relative to dry air, dimensionless

y = compressibility factor, dimensionless

ΔP = pressure drop across orifice, in. H_2O

The following values were either assumed or taken from the LEARY and TSAI report:

D_2 = 0.614 in

K = 0.60

G = 1.0

y = 1.0

P_o = 29.9 in. Hg (assumed)

T_o = 545° R (assumed)

With an inside pipe diameter of 3.068 in. and the use of the above values the air flow equation becomes:

$$\dot{m}_n = (60.5 \times 10^{-4})(1.0 - 0.00072\Delta P)\sqrt{\Delta P}$$

Where $\dot{m}_n$ is a nominal value based on the assumed orifice conditions. Actual airflow, $\dot{m}_a$, is given by

$$\dot{m}_a = \dot{m}_n\sqrt{545/T_o \times P_o/29.9}$$

For supercharged operation:

T_o = 545°R (assumed)

P_o = 39.9 in Hg (assumed)

Thus

$$\dot{m}_n = (70.3 \times 10^{-4})(1.0 - 0.00054\Delta P)\sqrt{\Delta P}$$

And

$$\dot{m}_a = \dot{m}_n\sqrt{545/T_o \times P_o/39.9}$$

Figures A-1 and A-2 show plots of $\dot{m}_n$ versus ΔP.

FIGURE A - 1

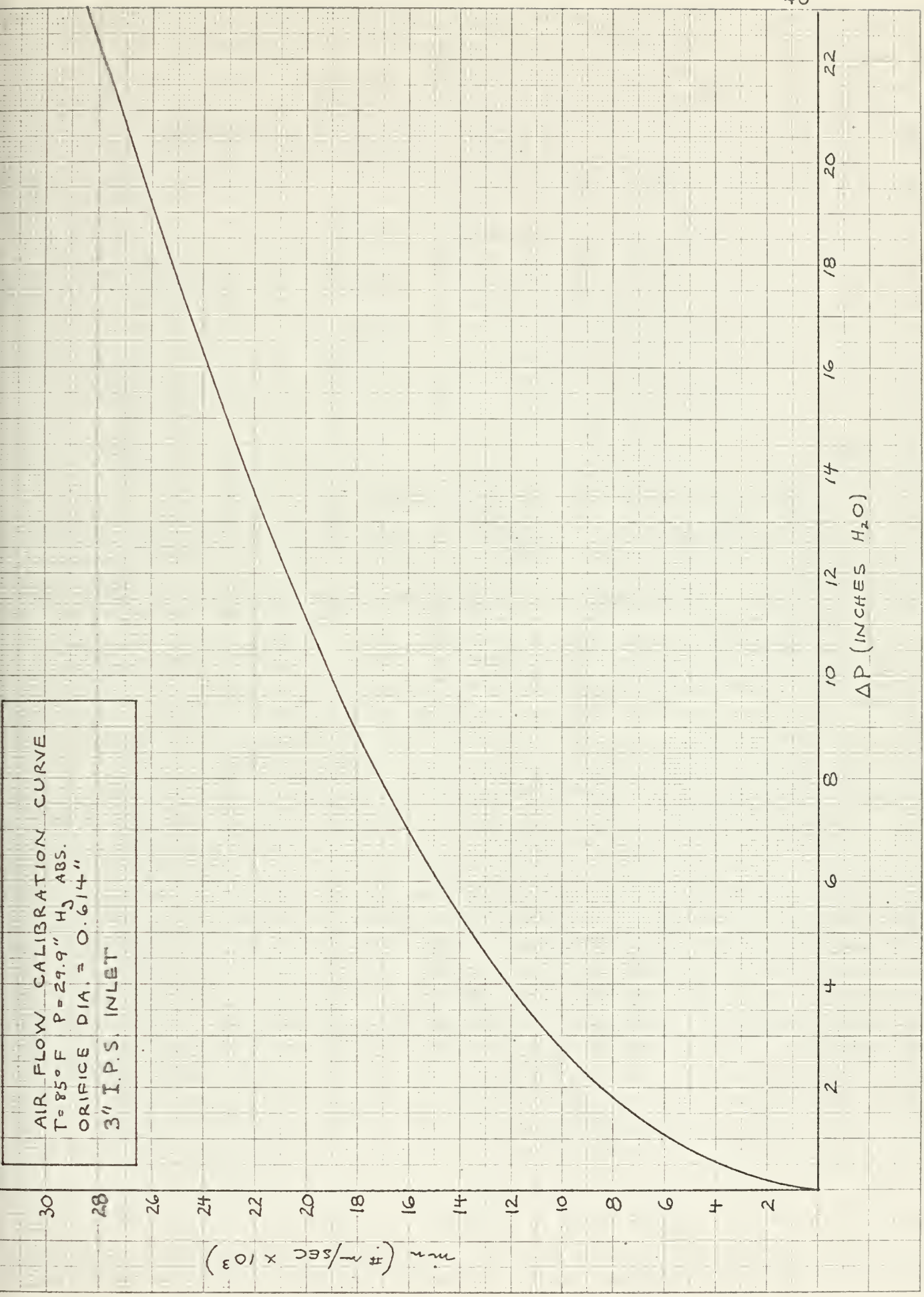
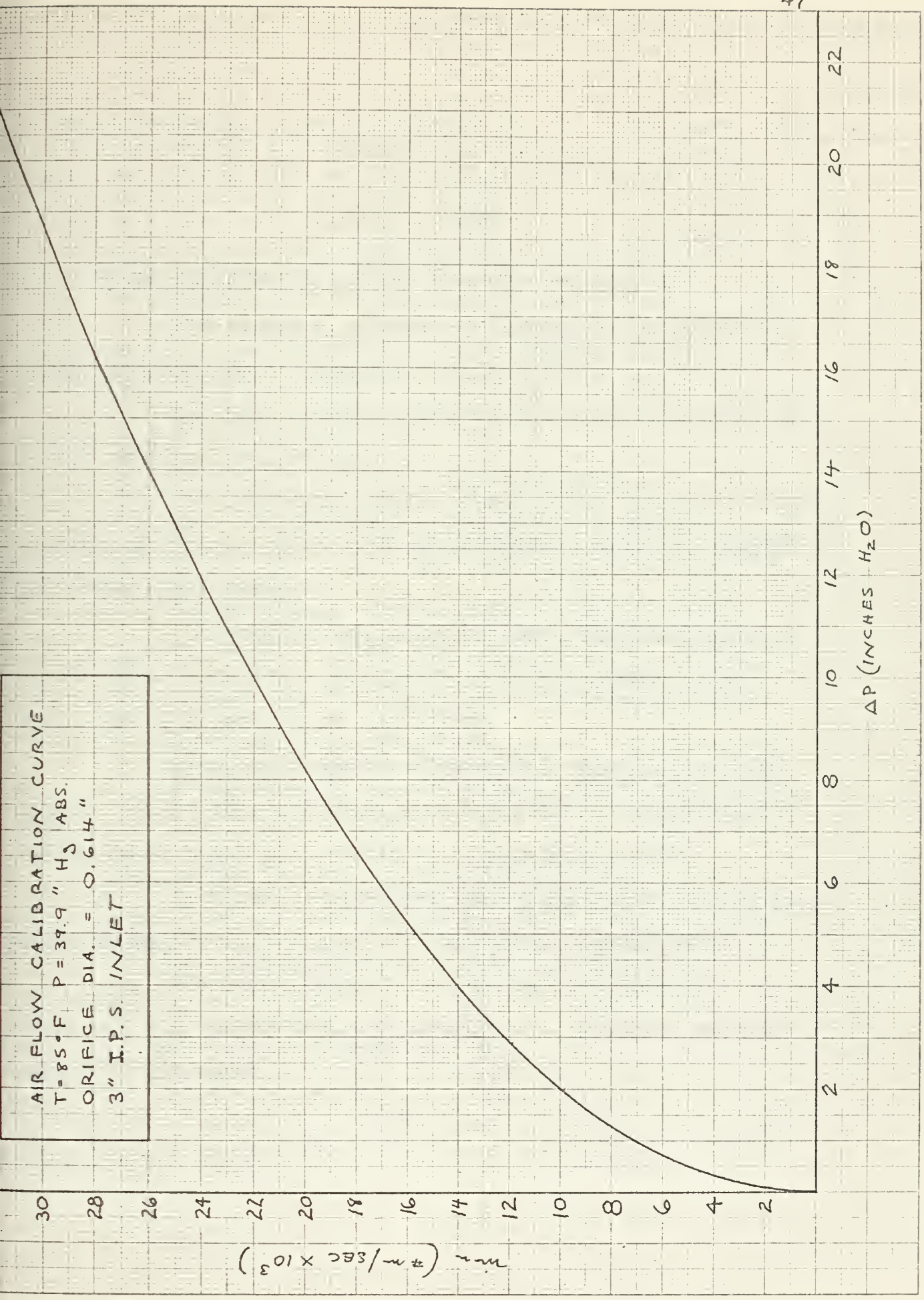

AIR FLOW CALIBRATION CURVE
T = 85° F P = 29.9" H_g ABS.
ORIFICE DIA. = 0.614"
3" I.P.S. INLET
$\dot{m}_a$ (# m/sec x 10^3)
30
28
26
24
22
20
18
16
14
12
10
8
6
4
2
ΔP (INCHES H_2O)
2
4
6
8
10
12
14
16
18
20
22

FIGURE A - 2


AIR FLOW CALIBRATION CURVE
T = 85°F P = 39.9 " H_3 ABS.
ORIFICE DIA. = 0.614"
3" I.P.S. INLET
$\dot{m}_a$ (#m/sec x 10^3)
ΔP (INCHES H_2O)
30
28
26
24
22
20
18
16
14
12
10
8
6
4
2
2
4
6
8
10
12
14
16
18
20
22

APPENDIX B

COMPUTER MODEL

A. A BRIEF DESCRIPTION OF THE COMPUTER PROGRAM

1. The program, written in FORTRAN IV, consists of a main program and a subprogram called GETVV.

2. The main program utilizes the equations of Chapter III in nondimensional form.

3. The subprogram GETVV computes the cylinder volume knowing the crank angle, cylinder diameter, conn rod length, and crank shaft offset.

4. The volumes and pressures are nondimensionalized with respect to initial conditions which are taken at the beginning of injection or soon after.

5. The heat transfer effects are computed in a DO loop. This DO loop performs an integration of a heat flux versus crank angle plot, using the trapezoidal rule.

6. The initial pressures and crank angles chosen for input should be at the start of injection or soon after.

7. The input parameters such as the molecular weight of air, gamma-air, etc. should be consistent with the region of interest.

B. DATA NECESSARY AS COMPUTER INPUT

1. ΔH at zero absolute temperature in units of joules/kilogram.*

2. Specific heat ratio of the burnt gases.*

3. Specific heat ratio of air.

4. Specific heat ratio of the fuel vapors.

5. Molecular weight of the air.

6. Molecular weight of the fuel.

7. Molecular weight of the burnt gases.

8. Weight of fuel burned per engine cycle in lb. mass/cycle.

9. Weight of air consumed per engine cycle in lb. mass/cycle.

10. Assumed fuel to air ratio of the burnt gases.

11. Mean fuel to air ratio.

12. Temperature of fuel vapors just after injection in ^{o}K.

13. Diameter of the cylinder in inches.

14. The crank to conn rod offset in inches.**

15. The distance from center to center of the connecting rod in inches.**

16. The engine compression ratio.

---

*See section C of this appendix for an explanation of how to obtain. **See Figure B - 2.

17. Closing of inlet valve in degrees measured from T.D.C.

18. Temperature at closing of the inlet valve in ^{o}K. (This is optional.)

19. The crank angles and their corresponding pressures in degrees of arc and lbs./ sq. in. respectively.

20. The heat flux corresponding to the crank angles of item 19, units are B.T.U's per ft^2. sec.*

21. The engine RPM.

C. CONSTRUCTION OF THE H VS. T GRAPH TO FIND THE VALUE OF Δ H

For combustion at constant enthalpy:

H_p = total enthalpy of products

H_r = total enthalpy of reactants

and

$$H_p = H_r$$

Expressing this in terms of combustion of air and fuel vapors to gaseous products:

$$H_r - H_r' = H_p - H_p' + H_{rp}'$$

Or based on a one pound mass of air:

$$(h_a - h_a') + F(h_f - h_f') + F(\Delta h_f') = (1 + F)(h_p - h_p')$$

$\Delta h_f'$ = lower heating value of the fuel. The subscripts a, f, and p refer to air, fuel vapor, and products respectively, and the primed values refer to some reference state.

---

*See Figure IV - 4.

Graphically the equation is shown in Fibure B - 1. Figure B - 2 illustrates items 14 and 15 of section B.

D. THE COMPUTER DATA DECK AND CARDS ARE ARRANGED AS FOLLOWS:

1. Card one, format (2I4,7F10.3)

a. The quantities on the first card are IDO, NOPTS, GAMB, GAMA, GAMF, EMWA, EMWF, EMWB, and RPM.

1. If IDO $>$ 0 the current case is computed, otherwise the program is terminated.

2. NOPTS is a number $\leq$ 50, which is the number of experimental data points (P vs. θ) supplied for this case.

3. GAMB, GAMA, and GAMF are the specific heat ratios of the burnt gases, air and fuel vapors, respectively.

4. EMWA, EMWF, and EMWB are the molecular weights of the air fuel, and products, respectively.

5. RPM is engine revolutions per minute.

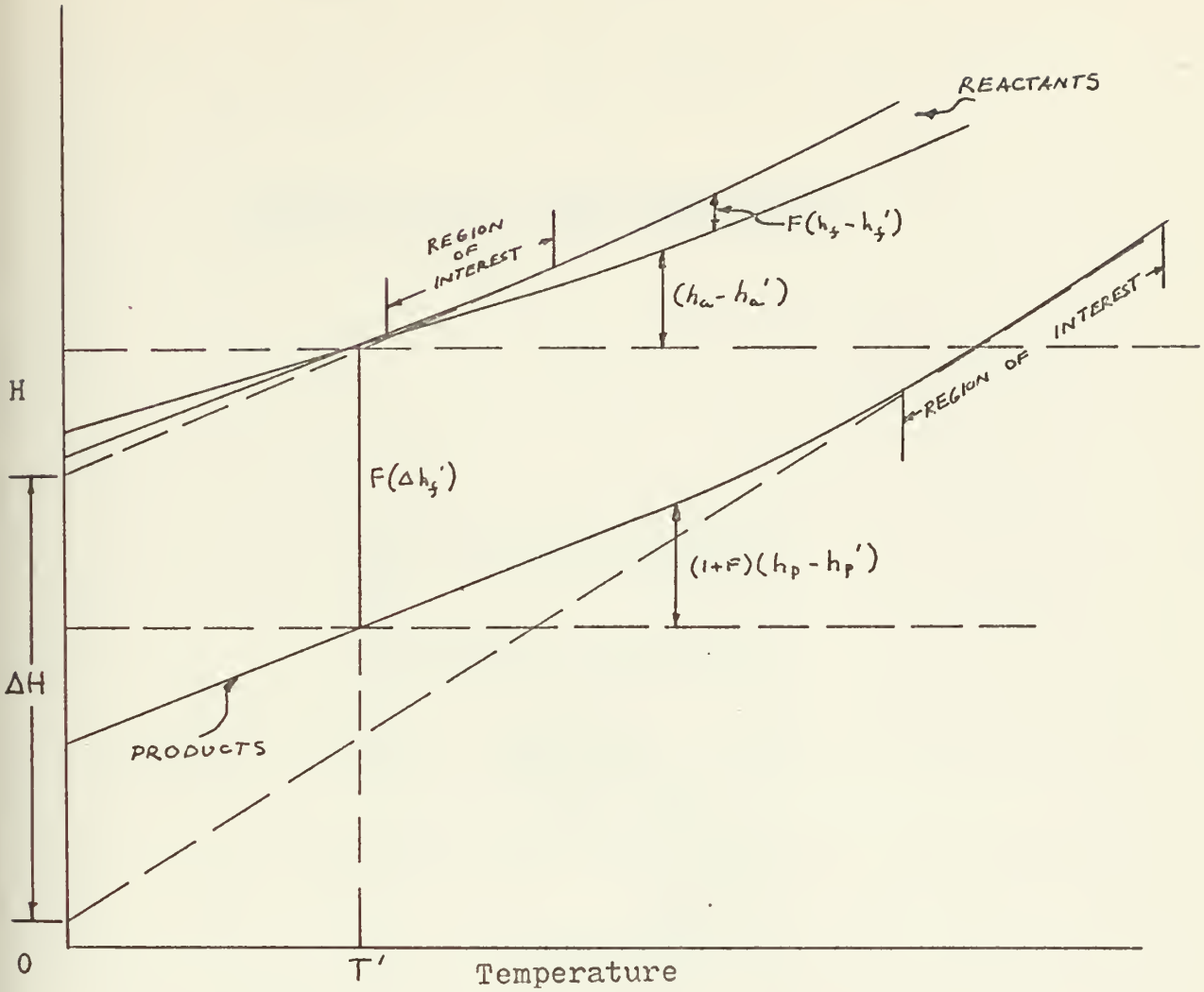
2. Card two, format (I4,7F10.3)

a. The quantities on this card are IDONT, DELH, WTFO, WTAO, FB, FO, TFO, and DIA.

1. If IDONT $>$ 0 the burned mixture calculations are omitted.

2. DELH is ΔH as determined from the H vs. T diagram.

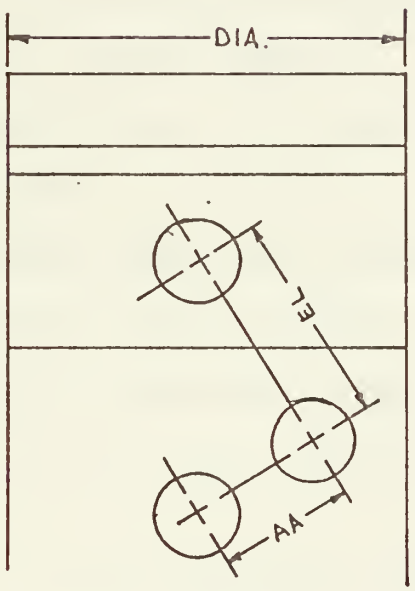
ENTHALPY VERSUS TEMPERATURE



$\triangle H$ is taken from the intercepts of the H axis at absolute zero temperature. The lines of interception are determined by fitting straight lines through the products and reactants curves over the regions of interest. The specific heat ratio of the burned gases is determined from the straight line fitted through the region of interest of the products curve, using the relationships $C_p = {}^{\Delta H}/_{\Delta T}$ and $\gamma_b = \frac{C_p}{C_p - R_b}$.

FIGURE B - 1

REQUIRED ENGINE INPUT DIMENSIONS



AA is the crank to conn rod offset and EL is the conn rod length.

FIGURE B - 2

3. WTFO, WTAO is the weight of fuel and air per cycle respectively.

4. FB is the assumed fuel to air ratio of the burnt gases.

5. FO is the mean fuel to air ratio.

6. TFO is the temperature of the fuel vapors at injection.

7. DIA is the cylinder diameter.

3. Card three, format (5E14.7)

a. The quantities on this card are AA, EL, CR, THTVC, and TPVC.

1. AA is the crank shaft offset. (Item 14)

2. EL is the conn rod length. (Item 15)

3. CR is the compression ratio.

4. THTVC is the angle where the inlet valve closes.

5. TPVC is the temperature when the inlet valve closes.

4. Card four and more, format (5E14.7)

a. The remaining cards are arranged in three sequences.

1. The first is for values of theta in degrees of arc.

2. The second is for values of pressure.

3. The third is for values of heat flux.

b. There are NOPTS values in each sequence and since the format is 5E14.7, there are no more than five values per card.

5. An option exists depending on the value of TPVC. If TPVC $>$ 0.0 the computer expects to encounter the card sequence for theta, pressure, and heat flux. However, if TPVC $\leq$ 0.0, the computer will attempt to calculate the value of TPVC. When this is the case more data must be supplied on a card inserted before the angle and pressure sequences. The quantities on this card are AMDOT, and EPS, with format 2F14.7.

a. AMDOT is the mass rate of flow of air in lbs. mass/sec.

b. EPS is the mole fraction of burned gases left in the cylinder at the start of the present combustion cycle.

c. This card should not be in the deck if TPVC $>$ 0.0.

E. PROGRAM OUTPUT

1. The first block of data is a listing of the input values.

2. If the computer was given the option of calculating TPVC, then it will print out a line headed by "computed". The outputs will be V_o in cubic centimeters, density of air (P_o) in grams per cubic centimeter, and TPVC in degrees Kelvin.

3. If the computer was not instructed to compute

TPVC, then the next line of output after the listing of the input values will be PVC, the closing pressure of the inlet valve in lbs/sq. in., TOA, the temperature of the air in degrees Kelvin at the initial data point, VO, the total volume of the air in cubic inches at the initial data point.

4. The next lines of output are listed by crank angle. For each angle listed there will be a P^*, the nondimensionalized pressure at that crank angle, V^*, the nondimensionalized total volume at that crank angle, a nondimensionalized work term, Q^*, the nondimensionalized heat transferred, integrated from the initial point up to the particular angle, and lastly Y, or the mass fraction of fuel burned at that crank angle. (Q is nondimensionalized with respect to PO * VO and the appropriate conversion factors.)

5. The next group of output values lists temperatures versus crank angle for the fully mixed case. TB is the mean temperature of the burnt gases, and TB(P) is the temperature of the burnt gases as calculated from the isenthalpic combustion.

6. The last group of output values lists the temperatures and pressures for the unmixed case. P Present is the pressure which the element is at, while P-Prime is the pressure at which the element burnt, and TB(PP,P) corresponds to the temperature if the element had burnt at PP and was isentropically compressed to P.

F. The following is a listing of the main program and subroutine GETVV.

```
      REAL NDIM,L1,L2
      DIMENSION AN(51),P(51),V(51),W(51),FLUX(51),QNDIM(51),Q(51)
      DIMENSION Y(51),TBP(51),TBST(51),TB(51),TBY(51),QST(51)
       EQUIVALENCE (VOVB,VOVB),(VOVC,VOVC), (VOCM,VOCM),(PO,PO)
      PI=3.141593
      RPD=PI/180.0
      R=8314.3
      AMDOT=0.0
      EPS=0.0
  100 CONTINUE
      READ(2,101) IDO,NOPTS,GAMB,GAMA,GAMF,EMWA,EMWF,EMWB,RPM
  101 FORMAT(2I4,7F10.3)
      IF(IDO)102,102,103
  102 CALL EXIT
  103 READ(2,301)  IDONT,DELH,WTFO,WTAO,FB,FO,TFO,DIA
  301 FORMAT( I4,7F10.3)
      READ(2,104) AA,EL,CR,THTVC,TPVC
  104 FORMAT(5E14.7 )
      IF(TPVC)142,142,143
  142 READ(2,105) AMDOT,EPS
  105 FORMAT(2F14.7)
  143 CONTINUE
      READ(2,104) (AN(IP),IP=1,NOPTS)
      READ(2,104) ( P(IP),IP=1,NOPTS)
      READ(2,104) (FLUX(IP),IP=1,NOPTS)
      WRITE(5,302) IDO,NOPTS,IDONT,DELH,GAMB,GAMA,GAMF,EMWA,EMWF,EMWB,
     1WTFO,WTAO,FB,FO,TFO,DIA,RPM
  302 FORMAT(12H1INPUT IDO =,I4,8H NOPTS =,I4,8H IDONT =,I4,7H DELH =,
     1E10.3,10H GAMMA-B =,E10.3,10H GAMMA-A =,E10.3,10H GAMMA-F =,E10.3
     2 / 2X,7H EMWA =,E10.3,7H EMWF =,E10.3,7H EMWB =,E10.3,7H WTFO =,
     3E10.3,7H WTAO =,E10.3 / 2X,5H FB =,E10.3,5H FO =,E10.3,6H TFO =,
     4E10.3,5H DIA=,E10.3,6H RPM =,E10.3)
      WRITE(5,303) AA,EL,CR,THTVC,TPVC
  303 FORMAT(4H A =,E10.3,4H L =,E10.3,5H CR =,E10.3,11H THETA-VC =,
     1 E10.3,7H T-VC =,E10.3  )
      WRITE(5,144) AMDOT,EPS
```

```
144   FORMAT(' ',6X,'AMDOT =',E10.3,6X,'EPS =',E10.3)
      WRITE(5,304) (AN(IP),IP=1,NOPTS)
 304  FORMAT(8H THETA =, 10(E10.3,1X)  )
      WRITE(5,305) (P(IP),IP=1,NOPTS)
 305  FORMAT(5X,3HP =, 10(E10.3,1X)   )
      WRITE(5,306) (FLUX(IP),IP=1,NOPTS)
306   FORMAT(5X,6HFLUX =, 10(E10.3,1X)   )
      GBM1=GAMB-1.0
      GAM1=GAMA-1.0
      GFM1=GAMF-1.0
      OOGA=1.0/GAMA
      OMOGA=1.0-OOGA
      OMOGB=1.0-1.0/GAMB
      OMOGF=1.0-1.0/GAMF
      DEL1=(GAMB-GAMF)/(GBM1*GFM1)
      DEL2=(GAMB-GAMA)/(GBM1*GAM1)
      RFUEL=R/EMWF
      RAIR=R/EMWA
      RBURN=R/EMWB
      DELHM=(DELH*WTFO*0.4536)
      CON2=EL*EL-AA*AA
      ELC=(AA+AA)/(CR-1.0)
      VCON1=ELC+AA+EL
      VCON2=ELC+AA+AA
      CALL GETVV(RPD,CON2,AA,AN(1),VCON1,VCON2,VV,ISENT)
      IF(ISENT)95,95,100
 95   VOVB=VV
      VBDC=0.25*DIA*DIA*PI*VCON2
      VO=VOVB*VBDC
      CALL GETVV(RPD,CON2,AA,THTVC,VCON1,VCON2,VV,ISENT)
      IF(ISENT)96,96,100
 96   VCVB=VV
      VCVO=VCVB/VOVB
      VOVC=1.0/VCVO
      PVC=P(1)*(VCVC**GAMA)
      PO=P(1)
```

```
      IF(TPVC)145,145,147
  145 VOCM=VOVB*VBDC*16.387
      PA=PO/14.7
      RHOA= (AMDOT*1.0E3/2.2046)/(.5*(RPM/60.0)*VOCM*(1.0-EPS))
      TPVC=(VOVC**GAM1)*(PA/RHOA)*(EMWA/82.0556)
      WRITE(5,146) VOCM,RHOA,TPVC
  146 FORMAT(16HOCOMPUTED VOCM =,E11.4,7H RHOA =,E11.4,7H TPVC =,E11.4)
  147 CONTINUE
      TERM1=(WTFO*0.4536*RFUEL*TFO)
      TERM2=(PO*VO*1.356)/12.0
      TOA=(TERM2-TERM1)/(0.4536*WTAO*RAIR)
      VOA=(0.4536*WTAO)*(RAIR*.7376*12.0)*(TOA)/PO
      DELHN=(DELHM*12.0)/(PO*VO*1.356)
      DO 106 I=1,NOPTS
      CALL GETVV(RPD,CON2,AA,AN(I),VCON1,VCON2,VV,ISENT)
      IF(ISENT)98,98,100
   98 V(I)=VV/VOVB
      P(I)=P(I)/PO
  106 CONTINUE
      W(1)=0.0
      DO 107 I=2,NOPTS
      DELW=.5*(P(I-1)+P(I))*(V(I)-V(I-1))
      W(I)=W(I-1)+DELW
  107 CONTINUE
      WRITE(5,97) PVC,TOA,VC,VOA
   97 FORMAT(5X,5HPVC =,E11.4,6X,4HTOA=,E11.4,2X,4HVO =,E11.4,2X, 9H
     1VOA =,E11.4)
      Q(1)=0.0
      QST(1)=0.0
      QNDIM(1)=0.0
      TIM=0.1668/RPM
      DO 120 I=2,NJPTS
      BASE=(AN(I)-AN(I-1))*TIM
      H1=FLUX(I-1)
      H2=FLUX(I)
      L1=(V(I-1)*VC*4.0)/(PI*DIA*DIA)
```

```
      L2=(V(I)*VO*4.0)/(PI*DIA*DIA)
      CONST=(PI*DIA*DIA)/2.0
      A1=CONST+(PI*DIA*L1)
      A2=CONST+(PI*DIA*L2)
      AM=(A1+A2)/2.0
      NDIM=(778.2*12.0)/(PO*VO)
      Q(I)=(BASE/2.0)*(H1+H2)*(AM/144.0)*(NDIM)
      QST(I)=Q(I)+QST(I-1)
      QNDIM(I)=FB*Q(I)+QNDIM(I-1)
  120 CONTINUE
      ILINE=2
      IX=0
      VOF=VO-VOA
      C1=VOF/VO
      C2=VOA/VO
      C3=RAIR/RBURN
      C4=(RFUEL/RBURN)*FB*(TFO/TOA)
      DO 113 I=1,NOPTS
      A=(P(I)*V(I)-1.0)/GBM1
      D1=(P(I)**OMOGF)
      D2=(P(I)**OMOGA)
      ENUM=FB*(A+W(I)+C1*DEL1*(D1-1.0)+C2*DEL2*(D2-1.0))
      DENOM=(FB*C1*DEL1*D1+FO*C2*DEL2*D2+DELHN)
      Y(I)=(ENUM+QNDIM(I))/DENOM
      TERM3=(C3*D2)/OMOGA
      TERM4=(C4*D1)/OMOGF
      TERM5=(DELH)/(RBURN*TOA)
      TBP(I)=OMOGB*(TERM3+TERM4+TERM5)
      ILINE=ILINE+4
      WRITE(5,115) AN(I),P(I),V(I),W(I),QST(I),Y(I)
  113 CONTINUE
  115 FORMAT(10HO  THETA =,E11.4,2X,4HP* =,E11.4,2X,4HV* =,E11.4,2X,8H
     1WORK =,E11.4,2X,4HQ* =,E11.4,2X,3HY =,E11.4)
      IF(IDONT) 116,116,100
  116 CONTINUE
      WRITE(5,117)
```

```
117  FORMAT(1H1,16HFULLY MIXED CASE)
     DO 200 I=1,NOPTS
     TBST(I)=TBP(I)
     IF(Y(I).LT.1.E-5) GO TO 200
     D3=C3*(P(I)**OMOGA)
     D4=C4*(P(I)**OMOGF)
     TERM6=(D3+D4)/(1.0+FB)
     D5=((P(I)*PO)*(V(I)*VO)*.25)/(WTAO*RBURN*TOA)
     D6=C3*(P(I)**OMOGA)+FC*(D4/FB)
     TERM7=(FB*(D5-D6))/(FC*Y(I)*(1.0+FB))
     TBST(I)=TERM6+TERM7
200  CONTINUE
     DO 205 I=1,NOPTS
     TB(I)=TBST(I)*TOA
     TBY(I)=TBP(I)*TOA
     WRITE(5,118) AN(I),TB(I),TBY(I)
205  CONTINUE
118  FORMAT(10HO  THETA =,E11.4,6X,4HTB =,E11.4,6X,7HTB(P) =,E11.4)
     WRITE(5,123)
123  FORMAT(1H1,12HUNMIXED CASE)
     DO 131 I=1,NOPTS
     PP=P(I)*PC
     JI=I
     DO 130 I1=1,JI
     PPR=P(I1)*PC
     TOUT=TBP(I1)*((PP/PPR)**OMOGB)
     TUMX=TOUT*TOA
     WRITE(5,132)PP,PPR,TUMX
130  CONTINUE
131  CONTINUE
132  FORMAT(9X,10HPPRESENT =,E11.4,3X,9HP-PRIME =,E11.4,2X,10HTB(PP,P
    1=,E11.4)
     GO TO 100
     END
```

```
      SUBROUTINE GETVV(RPD,CON2,AA,ANG,VCON1,VCON2,VV,ISENT)
      ISENT=0
      ARG=ANG*RPD
      ARG=AA* COS(ARG)
      DISC=ARG*ARG+CON2
      IF(DISC)100,100,102
  100 WRITE(5,101) ANG
  101 FORMAT(1X,22HDISC. NEG. FOR THETA =,E10.3,18H, GO TO NEXT CASE.  )
      ISENT=1
      GO TO 109
  102 DISC= SQRT(DISC)
      RT1=ARG+DISC
      RT2=ARG-DISC
      IF(RT1)106,108,103
  103 IF(RT2)108,108,104
  104 WRITE(5,105) RT1,RT2,ANG
  105 FORMAT(1X,4HR1 =,E10.3,5H R2 =,E10.3,8H THETA =,E10.3,
     1 19H.  GO TO NEXT CASE.  )
      ISENT=1
      GO TO 109
  106 IF(RT2)104,107,107
  107 RT1=RT2
  108 VV=(VCON1-RT1)/VCON2
  109 RETURN
      END
```

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